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ABSTRACT

Pressure pulsations inside the discharge system of fractional horse power reciprocating type compressor are often a source of objectionable noise or vibration in a refrigeration unit.. In this paper, modeling of the internal acoustic discharge manifold of the compressor has been performed by using 3 -D BEM model. The discharge muffler system was also instrumented for a bench test to verify the computational acoustic model. Results from the analysis are compared with those measured in experiments. One of the sound sources within the compressor is also the mechanical friction between the crankshaft thrust surface and facing surface of the outboard bearing. The sound radiated by the compressor has been reduced by use of the special designed thrust bearing which has been pressfit within countersunk recess in the rotor. This results in forming a single frictional pair, the lower surface of the thrust bearing against the upper end face of the compressor main bearing hub, thereby reducing overall sound radiated by the compressor as during operation, so at the start.

INTRODUCTION

The reciprocating fractional horse power compressors are widely used in refrigeration. However vibration and noise characteristics of such compressors required further improvement to compete with rotary compressors which have better balanced motor, lower discharge and suction gas pulsation, and absence of dynamic valve on suction side. Small refrigeration compressor generally includes hermetic housing, electric motor at the top, and compression mechanism at the bottom as shown in Fig.1. The compressor is usually spring mounted inside the housing to prevent most of the compressors vibration from being transferred outside. Electric motor consists of stator and rotor secured to the crankshaft. The weight of the rotor and crankshaft combination is generally supported by the thrust bearing located at the top of the hub. The discharge line inside the housing is made from small diameter tubing which connect compressor mechanism inside the housing with the refrigeration system. In the case of reciprocating piston type compressor, the compression mechanism itself is the principal source of noise. The compression mechanism induces noise and vibrations by the periodic change of the gas compression moment, mechanical forces, and fluctuation of the electric motor torque. The refrigerant gas pulsations takes place on both the low and high pressure sides, but the suction pressure pulsations are suppressed to some degree by large internal volume of the housing and suction muffler. The discharge gas pulsations not only trigger resonance of the discharge system but also transfer structural vibration to the housing , and trigger vibrations and following sound radiation of the connected refrigeration system [1,2]. In this paper the fluid-borne noise problem was studied in some details by analytical evaluation of the internal acoustic characteristics of the discharge system with help of I-Deas software and experimental verification tests of the real compressor discharge system to support analytical findings.

One area that has received attention is that of mechanical friction within the compressor [3]. The crankshaft of compressor generally rotates about a vertical axis and therefore requires journalling within a frame and bearings to position and confine its rotation. A thrust bearing is used to bear the weight of crankshaft and motor parts. In the past, these bearings have generally been of a plain or oil film type, either having two machined surfaces rubbing together or one or two hardened washers. Another type of bearing in use is a ball bearing system. A disadvantage of ball bearings is that they increase the noise of the compressor and also increase the cost of the compressor as well.

DISCHARGE SYSTEM

The discharge muffler system consists a pair of chambers formed on the lower side of the cylinder block (see Fig.1). The discharge gases flow from the discharge plenum cavity in the cylinder head through passage to he chamber #1 in the cylinder block. Each of the muffler chambers have a hemispherical cap bolted in place. A transfer tube extends between two hemispherical caps to conduct the discharge gas from the chamber #1 into the chamber #2. A second tube then extends from the cap on the chamber #2 through the necessary convolutions to allow flexing, and to the exterior of the housing. The combination of these two discharge cavities provide a high degree of volumetric efficiency while retaining multiple chamber filters which allow some degree of sound reduction so that compressor can operate as quietly as possible.

Acoustic measurements performed in the anechoic room with small reciprocating compressor shows that maximum sound level peaks have been observed at 500 Hz and in the frequency range 2.5 kHz - 8 kHz.

A 3-D model of the discharge flow was created for internal acoustic analysis. The model shown in Fig.2 represents the volume bounded by the inner surfaces of the cylinder head, crankcase cavities with spherical caps which are exposed to discharge fluid. The discharge valve stop and region around the stop were also included. The model was created from parosolid files for the baseline analysis using 15,000 tetrahedron elements and 23,000 nodes. The discharge fluid used in the model of the entire discharge flow path was refrigerant R-134A at the discharge conditions of the

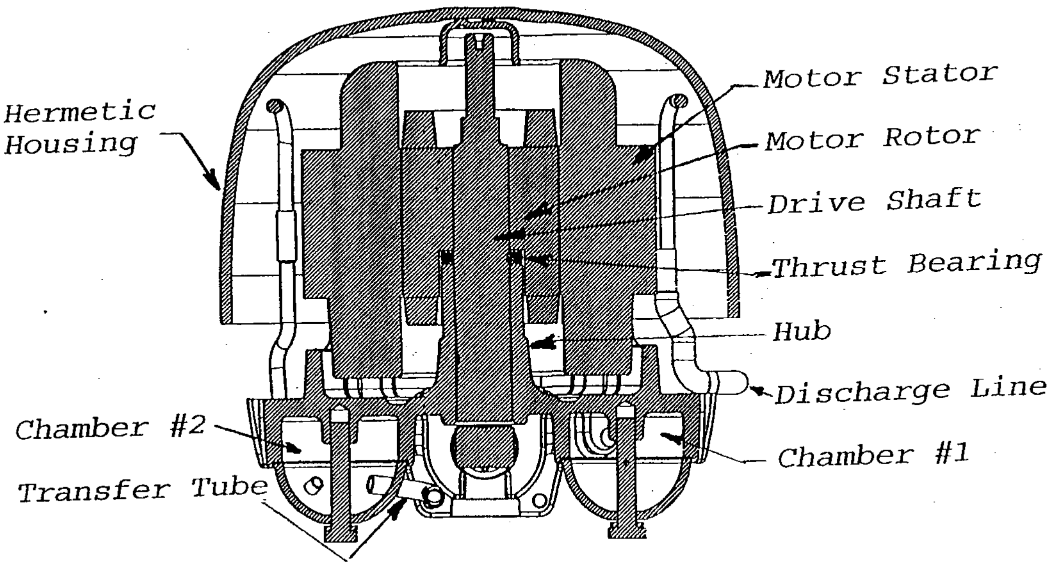

Fig.1.Cross-sectional view of refrigeration compressor

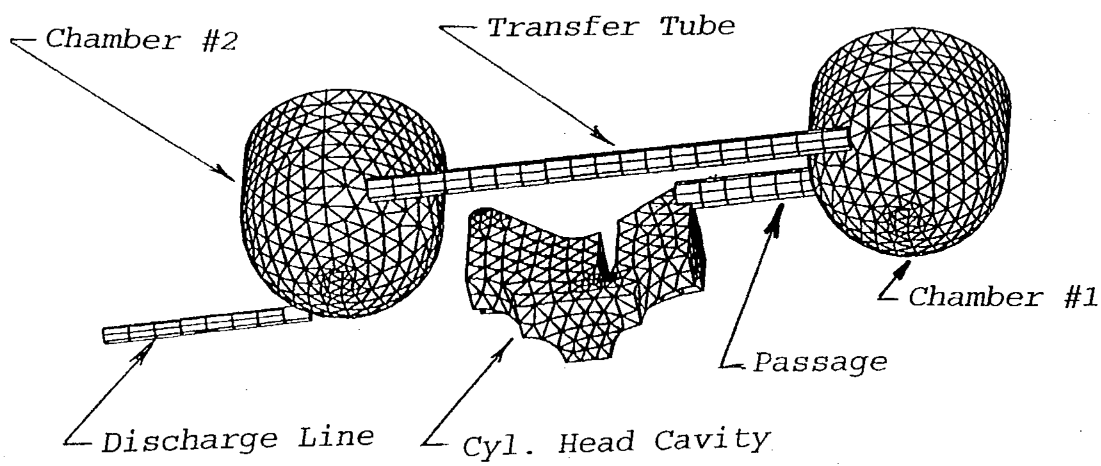

Fig.2. Acoustic Mesh for the Discharge System
Internal Structure

operating compressor. A mode analysis performed for the muffler system helped to reveal the acoustic modes which are shown in Table 1.

The end of the transfer tube located in chamber#2 (inlet) and outlet of the chamber #2 are 180° apart and positioned almost on the same level. The positioning of inlet and outlet has been primarily for convenience of construction and adaptation to the size of compressors, as opposed to obtaining maximum sound attenuation. Generally , the pressure distribution in a circular volume is proportional to $\mathbf{J}_m(x)$ where $\mathbf{J}_m$ is the Bessel Function of the first kind of order m and $\mathbf{x}$ is proportional to radius. So it is possible to achieve substantially greater sound attenuation by precisely locating the inlet and outlet. In the analytical study two design iteration along with the baseline were evaluated for the discharge side modification. The first design modification involves mounting disposition of inlet to the chamber #2 circumferentially to have 90° between inlet and outlet [4].

Table 1

Mode #	*Freq., Hz*	*Element*	*Mode #*	*Freq., Hz*	*Element*
1	75.3	Cyl.Head	10	2292	Chamber
2	187	Passage	11	2442	Cyl.Head
3	816	Passage	12	2447	Transfer Tube
4	958	Cyl.Head	13	2561	Chamber
5	1620	Passage	14	2592	Chamber
6	1738	Transfer Tube	15	3267	Chamber
7	2244	Chamber	16	3355	Transfer Tube
8	2269	Chamber	17	3599	Chamber
9	2270	Chamber			

In the second change the output end of the passage connecting cavity of the cylinder head with chamber #1 has been located at the middle of the cavity in vertical direction. The input end to the chamber #1 has been extended to the vertical axis of the first chamber. Analytical FRF were synthesized for these modifications and compared to the base line spectra (see **Fig. 3A** and **3B**). The acoustic FE model confirms the presence of several modes in the cylinder head cavity and discharge system first and second chambers. Analytical study indicate that specified above modifications can reduce up to 20 dB sound peaks in the frequency range 2,000Hz-5,000Hz. The experimental test of small reciprocating compressor in which the inlet and outlet of the discharge system are located 90° from each other in both the first and secondary chambers has been performed in anechoic chamber of Tecumseh Products Sound Department. Results of the experimental study shown in Fig.4 indicate up to 2 dBA reduction of overall sound. The modification affected the frequency range

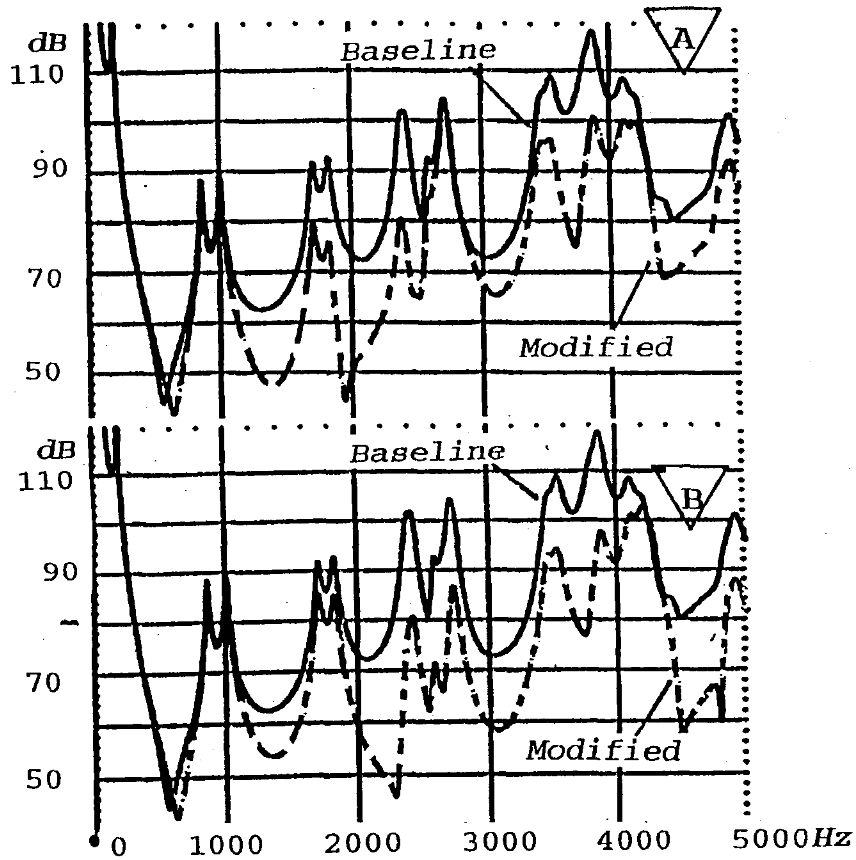

Fig.3. Response of the Discharge System:
A - Input and Output Offset 90°.
B - Extended passage between cylinder head cavity and first muffler chamber.

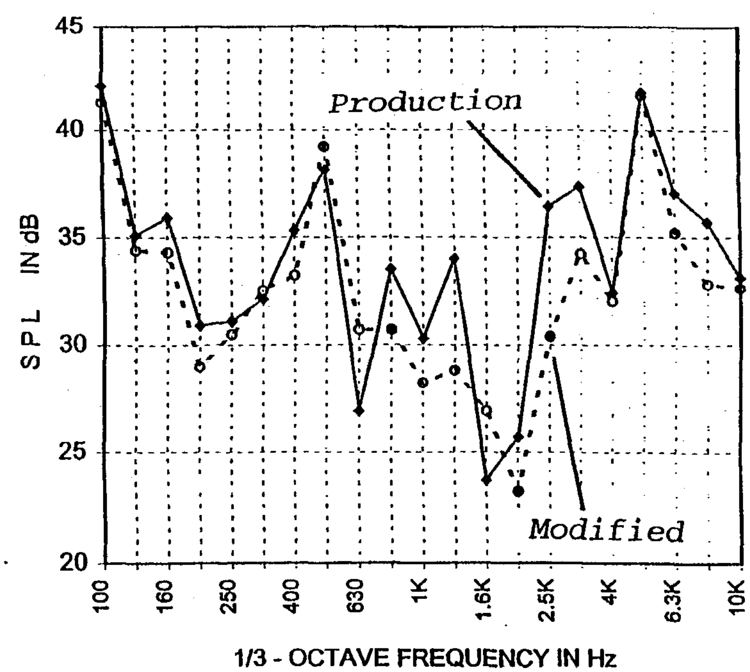

Fig.4.Effect of the modification on the compressor noise

which has been predicted by analytical study.

THRUST BEARING

In compressor application due to dilution of refrigerant mixed with lubricant, and due to high temperatures, the viscosity of the lubricant mixture becomes so low that using a thrust bearing at the crankshaft thrust face is helpful in reducing wear. The mechanical friction between the crankshaft thrust surface and facing surface of the outboard bearing is one of the noise sources. Noise produced by such a hydrodynamic bearing become significant when a full oil film is not generated or when the bearing operating conditions are such that the self-generated instability known as oil whirl occurs. Due to the limited space the thrust area is relatively small. It creates conditions for partial or total overloading of the bearing.
The total axial force **F** applied to the thrust surface

$$\mathbf{F} = \mathbf{F_M} + \mathbf{F_R} + \mathbf{F_C}, \qquad (1)$$

where $\mathbf{F_M}$ is the motor axial (solenoid) force, $\mathbf{F_R}$ and $\mathbf{F_C}$ correspondingly is gravity force of the rotor and crankshaft. The motor axial (solenoid) force can be computed from the equation below [5]:

$$\mathbf{F_M = 0.0117P \ (60/f) \ (I_{M0}\, E_0 / L_0) \ (L_0/L)^2 \, [1 - 2\pi^{-1} ctn^{-1} (h/g)]} \qquad (2)$$

where **P** - phase number (for single phase =2), **f** - line frequency, $\mathbf{I_{M0}}$ - magnetizing current in amperes, $\mathbf{E_0}$-line voltage, $\mathbf{L_0}$- stator core stock height, **L** - effective core height, **h** - misalignment, and **g** - rotor-stator air gap. Another factors which significantly affect performance, radiated sound, and reliability are metal to metal contact due to poor oil film generation caused by saturation of the refrigerant in the oil (holes in the oil film) . The dynamic of the thrust bearing during start and operation of the compressor is governed by the torques exerted on it. Since the configuration of this thrust bearing is a parallel face, a geometric converging wedge for fluid friction is not shaped. The boundary friction loss F_L is

$$\mathbf{F_L = 2\mu \, W_S \, (R_{S2}^3 - R_{S1}^3) / 3 \ (R_{S2}^2 - R_{S1}^2)} \qquad (3)$$

where μ - coefficient of friction $\mathbf{R_{S1}}$ and $\mathbf{R_{S2}}$ - inside and outside radius of the thrust surface $\mathbf{W_S}$ - weight of rotor and shaft. With the addition of the axial solenoid downward force the loss factor will be significantly higher. It can cause conditions at which the oil film breaks down so that the metal-to-metal contact and imperfect lubrication begin. The mechanical friction associated with a vertical rotor and crankshaft combination as it rests upon and rotates about the frame bearing hub is reduced both at start up and during the compressor operation by utilizing a thrust bearing formed of a polyamide material [6]. By press fitting the thrust bearing within the counterbore formed in the rotor (see Fig. 5), rotation of the thrust bearing relative to the rotor is prevented [7]. This results in rotational contact between a single frictional pair, the lower surface of the thrust bearing against the upper end face of the bearing hub, thereby reducing the amount of mechanical friction loss within the

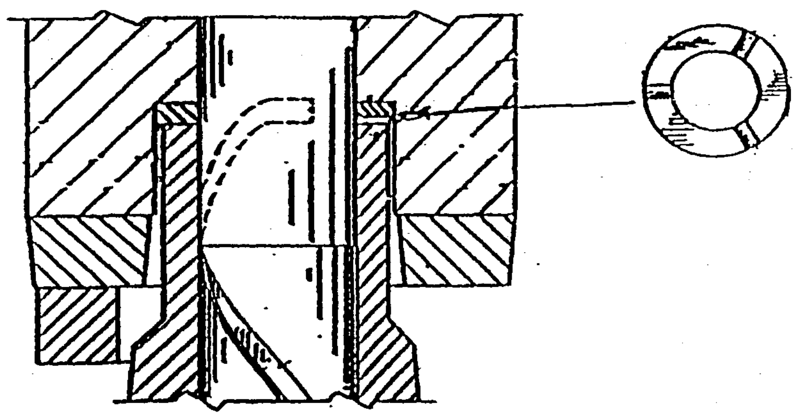

Fig.5.Cross-Sectional View of the Rotor, Bearing Hub and Thrust Bearing

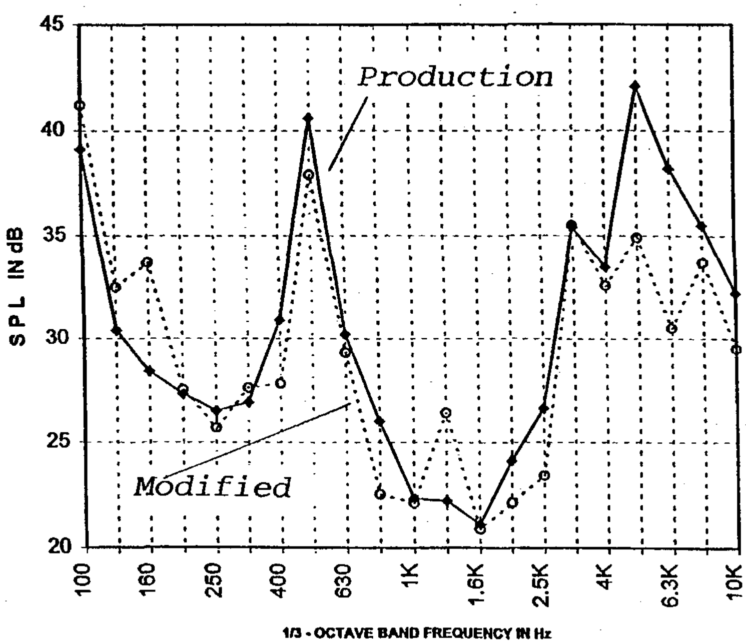

Fig.6.Effect of the Vespel Thrust Bearing on the Compressor Noise

compressor. In the preferred embodiment (see Fig. 5), the polyamide thrust bearing is formed of torlon as produced by Amoco or Vespel as produced by DuPont. By reducing the friction caused by the radial reaction of the crankshaft at compressor start up and during operation, the present modification increases overall compressor efficiency and reduces radiated sound. The polyamide material used to form the thrust bearing is characterized by a very low coefficient of static and kinetic friction. This results in reduced mechanical friction and reduced power consumption associated with starting and operation of the compressor. Another beneficial characteristic associated with polyamide is it's broad temperature range thermal stability. Even unlubricated polyamide thrust bearings are capable of withstanding approximately 300,000 lb. ft/in. minimum with a maximum contact temperature of 740° F. Lubrication oil is delivered by the crankshaft to the thrust bearing surface, thereby further reducing the coefficient of friction during compressor operation. Circular shape of new thrust bearing helps to form circumferential periodic pattern of the oil film. The consequence of the flow pattern in the bearing is extremely important to the rotor stability. When an oil flow has a circumferential pattern it generates a dynamic effect which creates rotating forces that, in feedback, act on the shaft and cause lateral precession motion.
New thrust bearing helps to eliminate occurrence of self-exited vibrations associated with such phenomena as oil whirl which triggered by fluid dynamic forces generated in the bearing. Yet additional advantages of the bearing relocation and modification are: vibration dampening, lack of corrosion, broad temperature range thermal stability, and superior chemical and abrasion resistance. Results of the experimental study shown in Figure 6 indicate up to 2 dBA reduction of overall sound.

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